Notice No.10

Rules and Regulations for the Classification of Special Service Craft

July 2017

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: June 2018

Amendments to	Effective date	IACS/IMO implementation (if applicable)
Part 9, Chapter 1, Section 4	1 July 2018	1 July 2018
Part 9, Chapter 2, Section 9	1 July 2018	1 July 2018
Part 10, Chapter 1, Sections 3 & 11	1 July 2018	1 July 2018
Part 10, Chapter 1, Sections 1, 7, 8, 9 & 11	1 July 2018	N/A
Part 16, Chapter 2, Sections 1, 9 & 11	1 July 2018	N/A

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Part 9, Chapter 1
General Requirements for Machinery

■ *Section 4*
Operating Conditions

4.7 Astern power

~~4.7.1~~ ~~Sufficient astern power is to be provided to maintain control of the ship in all normal circumstances.~~

~~4.7.2~~ ~~Astern turbines are to be capable of maintaining in free route astern 70 per cent of the ahead revolutions, corresponding to the maximum propulsion shaft power for which the machinery is to be classed, for a period of at least 30 minutes without undue heating of the ahead turbines and condensers.~~

4.7.1 In order to maintain sufficient manoeuvrability and secure control of the ship in all normal circumstances, the main propulsion machinery is to be capable of reversing the direction of thrust so as to bring the ship to rest from the maximum service speed. The main propulsion machinery is to be capable of maintaining in free route astern at least 70 per cent of the ahead revolutions corresponding to the maximum continuous ahead power for which the vessel is classed.

Part 9, Chapter 2
Surveys During Construction, Installation and Sea Trials

■ *Section 9*
Sea trials

9.2 Programme

9.2.2 Main propulsion systems are to undergo tests to demonstrate the astern response characteristics. The tests are to be carried out over at least the manoeuvring range of the propulsion system and from all control positions. A test plan is to be provided by the yard and accepted by the Surveyor. If specific operational characteristics have been defined by the manufacturer, then these are to be included in the test plan.

Existing paragraph 9.2.2 has been renumbered 9.2.3.

9.3 Performance testing

9.3.2 The reversing characteristics of the propulsion plant, including the blade pitch control system of controllable pitch propellers, are to be demonstrated and recorded during trials.

Existing paragraphs 9.3.2 and 9.3.3 have been renumbered 9.3.3 and 9.3.4.

~~9.3.4~~ 9.3.5 ~~Overloading of machinery is not to occur under continuous astern power.~~ For main propulsion systems with reversing gears, controllable pitch propellers or electric propeller drive, running astern is not to lead to the overload of propulsion machinery.

Part 10, Chapter 1
Reciprocating Internal Combustion Engines

■ *Section 1*
General requirements

1.1 ~~Scope~~ Application

1.1.1 The requirements of this Chapter are applicable to reciprocating internal combustion engines operating on liquid, gas or dual fuel (hereinafter referred to as engines) for main propulsion and essential auxiliary services. *Pt 10, Ch 1, 3 Crankshaft design* is not applicable to auxiliary engines having powers of less than 110 kW.

1.1.2 Engines providing power for services essential to the safety of the vessel are to be constructed under survey and in accordance with the requirements of this Chapter (*see also Pt 1, Ch 2, 3.9 Class notations (machinery) 3.9.1*).

~~1.1.7~~ 1.1.3 This Chapter is to be read in conjunction with the General Requirements for Machinery in *Pt 9 General Requirements for Machinery*.

~~1.1~~ 1.2 Scope

Existing paragraphs 1.1.3 to 1.1.6 have been renumbered 1.2.1 to 1.2.4.

~~1.2~~ 1.3 Approval process

Existing paragraph 1.2.1 has been renumbered 1.3.1.

~~1.2.2~~ 1.3.2 Each complete engine, as defined in ~~the scope~~ *Pt 10, Ch 1, 1.2 Scope*, intended for installation on an LR Classed vessel, is to have an LR Engine Certificate.

Existing paragraph 1.2.3 has been renumbered 1.3.3.

(Part only shown)
~~1.2.4~~ 1.3.4 To apply for an LR Engine Certificate, the following are to be submitted:

(a) a list of all documents identified in the ‘for information’ and ‘for appraisal’ columns of *Table 1.1.1 Plans and particulars to be submitted* with the relevant drawing numbers and revision status. This list is to cross-reference the approved plans previously submitted ~~in accordance with~~ ~~*Pt 10, Ch 1, 1.2 Approval process 1.2.4.(a)*~~ as part of the engine Type Approval and identify any plans that have been modified.

(b) where there is a licensor/licensee arrangement the list required by *Pt 10, Ch 1, 1.2 Approval process 1.3.4(a)* is to cross-reference the drawings submitted by the designer ~~in accordance with~~ ~~*Pt 10, Ch 1, 1.2 Approval process 1.2.1*~~ as part of the engine Type Approval. This list is to identify all changes where the approved design has been modified by the licensee. Where the licensee proposes design modifications to components, a statement is to be made confirming the licensor’s acceptance of the proposed changes. If designer/licensor’s acceptance is not confirmed, the engine is to be regarded as a different type and is subject to the complete appraisal and type approval process.

Existing paragraph 1.2.5 has been renumbered 1.3.5.

~~1.2.6~~ 1.3.6 For appraisal of emergency generator~~s~~ engines and turbochargers additional submissions are required. *See Pt 10, Ch 1, ~~1.3~~ 1.4 Submission requirements ~~1.3.4~~ 1.4.4* and *Pt 10, Ch 1, ~~1.3~~ 1.4 Submission requirements ~~1.3.5~~ 1.4.5* as applicable.

~~1.3~~ 1.4 Submission requirements

~~1.3.1~~ 1.4.1 The plans and information are to be submitted as required in ~~*Pt 10, Ch 1, 1.3 Submission requirements 1.3.1*~~ *Table 1.1.1 Plans and particulars to be submitted* and *Pt 10, Ch 1, ~~1.3~~ 1.4 Submission requirements ~~1.3.2~~ 1.4.2* to *Pt 10, Ch 1, ~~1.31~~ 1.4 Submission requirements ~~1.3.8~~ 1.4.8* as applicable.

(Part only shown)
~~1.3.2~~ 1.4.2 A schedule of testing at engine ~~builders~~ packager’s or system integrator’s facility, pre-sea trial commissioning and sea trials is to be submitted. The test schedules are to identify all modes of engine operation and the sea trials are to include typical port manoeuvres under the intended engine operating modes. The schedule is to include:

Existing paragraphs 1.3.3 to 1.3.8 have been renumbered 1.4.3 to 1.4.8.

Existing sub-Section 1.4 has been renumbered 1.5.

■ *Section 3*

Crankshaft design

Existing Section 3 has been deleted in its entirety and replaced with below.

3.1 Application

3.1.1 *Pt 10, Ch 1, 3 Crankshaft design* is not applicable to auxiliary engines having powers of less than 110 kW.

3.2 Scope

3.2.1 The formulae given in this Section are applicable to solid or semi-built crankshafts of forged or cast steel, having a main support bearing adjacent to each crankpin.

3.2.2 This section uses the static determinate method; alternative methods, including a fully documented stress analysis, will be specially considered.

3.2.3 Calculations are to be carried out for the maximum continuous power rating for all intended operating conditions.

3.2.4 Designs of crankshafts not included in this scope will be subject to special consideration.

3.2.5 Where a crankshaft design involves the use of surface treated fillets, or when fatigue parameter influences are tested, or when working stresses are measured, the relevant documents with calculations/analysis are to be submitted to LR.

3.2.6 The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas. The calculation is also based on the assumption that the areas exposed to highest stresses are :
(a) fillet transitions between the crankpin and web as well as between the journal and web,
(b) outlets of crankpin oil bores.

3.2.7 When the journal diameter is equal to or larger than the crankpin one, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate documentation of fatigue safety will be specially considered.

3.2.8 Calculation of crankshaft strength consists initially in determining the nominal alternating bending (*see Pt 10, Ch 1, 3.6 Calculation of bending stresses*) and nominal alternating torsional stresses (*see Pt 10, Ch 1, 3.7 Calculation of torsional stresses*) which, multiplied by the appropriate stress concentration factors (SCF) (*see Pt 10, Ch 1, 3.8 Stress concentration factors*), result in an equivalent alternating stress (uniaxial stress) (*see Pt 10, Ch 1, 3.9 Equivalent alternating stress*). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (*see Pt 10, Ch 1, 3.11 Fatigue strength*). This comparison will show whether or not the crankshaft concerned is dimensioned adequately (*see Pt 10, Ch 1, 3.12 Acceptability criteria*).

3.2.9 Further information and guidance for crankshaft design is provided in the LR guidance notes *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* and *Guidance for the Evaluation of Crankshaft Fatigue Tests.*

3.3 Information to be submitted

3.3.1 For the calculation of crankshafts, the documents and particulars listed below are required, this information is provided by completing LR Form 2073 and submitting the applicable plans required in *Table 1.1.1 Plans and particulars to be submitted*:

- Crankshaft drawing (which must contain all data in respect of the geometrical configurations of the crankshaft);
- Type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod);
- Operating and combustion method (2-stroke or 4-stroke cycle/direct injection, precombustion chamber, etc.) Number of cylinders;
- Number of cylinders;
- Output power at maximum continuous rating (MCR), in kW;
- Output speed at maximum continuous power, in rpm;
- Maximum firing pressure, $P_{\max}$, in MPa;
- Mean indicated pressure, in MPa;
- Charge air pressure (before inlet valves or scavenge ports, whichever applies), in MPa;
- Digitised gas pressure/crank angle cycle for MCR (presented at equidistant intervals at least every 5° CA);
- Mean piston speed;
- Compression ratio;
- Vee angle α_v, in degrees;
- Firing order numbered from driving end, *see Figure 1.3.1 Designation of cylinders;*
- Direction of rotation;
- Cylinder diameter, in mm;
- Piston stroke, in mm;
- Centre of gravity of connecting rod from large end centre, in mm;

- Radius of gyration of connecting rod, in mm;
- Length of connecting rod between bearing centres, L_H, in mm;
- Mass of single crankweb (indicate if webs either side of pin are of different mass values), in kg;
- Centre of gravity of crankweb mass from shaft axis, in mm;
- Mass of counterweights fitted (for complete crankshaft) indicate positions fitted, in kg;
- Centre of gravity of counterweights (for complete crankshaft) measured from shaft axis, in mm;
- All individual reciprocating masses acting on one crank, in kg;
- Crankshaft material specification(s) (according to ISO, EN, DIN, AISI, etc.);
- Mechanical properties of material (minimum values obtained from longitudinal test specimens):
  - tensile strength, in N/mm²
  - yield strength, in N/mm²
  - reduction in area at break, percentage
  - elongation, percentage
- Method of manufacture (free form forged, continuous grain flow forged, drop-forged, etc.; with description of the forging process);
- For semi-built crankshafts – minimum and maximum diametral interference, in mm;
- Particulars of alternating torsional stress calculations, *see Pt 10, Ch 1, 3.7 Calculation of torsional stresses*.

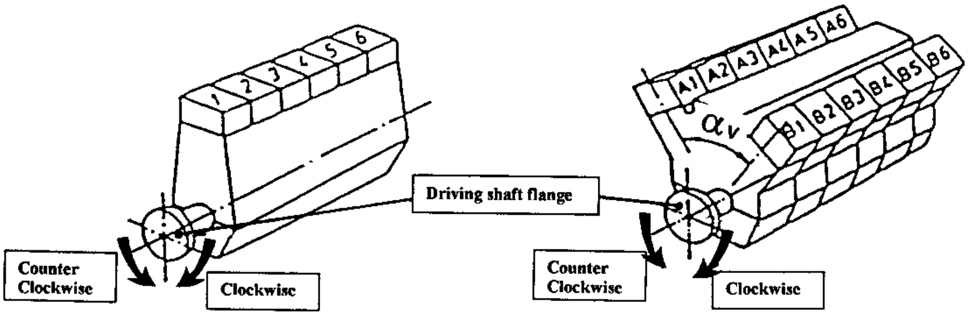


Figure 1.3.1 Designation of cylinders

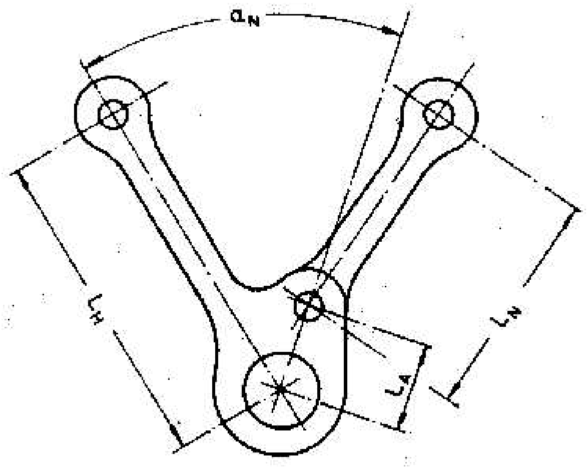


Figure 1.3.2 Articulated-type connecting rod

3.3.2 The following information is also required for appraisal of the crankshaft (not contained in Form 2073):

- For engines with articulated-type connecting rod (*see Figure 2.3.2 Articulated-type connecting rod*):
  - Distance to link point L_A, in mm
  - Link angle α_N, in degrees
  - Connecting rod length L_N, in mm
- firing interval (if applicable) i.e. if not evenly distributed
- Mass of connecting rod (including bearings), in kg;
- Mass of piston (including piston rod and crosshead where applicable), in kg;
- Every surface treatment affecting fillets or oil holes shall be specified so as to enable calculation according to Chapter 2 of the LR *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method;*
  - This is to include Crankshaft fatigue enhancement factors K_1 and K_2 where applicable.
- Maximum alternating torsional stress τa (N/mm^2)
- Mechanical properties of material (minimum values obtained from longitudinal test specimens), in addition to the information listed above:

Impact energy K_V, in Joules

3.4 Symbols

3.4.1 For the purposes of this Chapter the following symbols apply, *see also Figure 1.3.3 Crank dimensions for overlapped crankshaft*, *Figure 1.3.4 Crank dimensions for crankshaft without overlap*, *Figure 1.3.5 Crankpin section through the oil bore* and *Figure 1.3.6 Crankthrow of semi-built crankshaft*:

B = transverse breadth of web, in mm

D = crankpin diameter, in mm

D_{A} = the outside diameter of web or twice the minimum distance between centre-line of journals and outer contour of web, whichever is less, in mm

D_{BH} = diameter of axial bore in crankpin

D_{BG} = diameter of axial bore in journal

D_{G} = journal diameter

D_{o} = diameter of radial oil bore in crankpin, in mm

D_{s} = shrink diameter of main journal in web, in mm

E = pin eccentricity

E_{m} = Young's modulus of crankshaft material, in N/mm^2

F = area related to cross-section of web, in mm^2

K_{e} = bending stress factor (considers the influence of adjacent crank and bearing restraint)

K = fatigue enhancement factor ($K = K_1.K_2$)

K_1 = fatigue enhancement factor due to manufacturing process

K_2 = fatigue enhancement factor due to surface treatment
L_{s} = length of shrink fit, in mm

M_{BON} = alternating bending moment calculated at the outlet of crankpin oil bore

M_{BRFN} = alternating bending moment related to the centre of the web, in Nm

M_{TN} = maximum alternating torque, in Nm

M_{Tmax} = maximum value of the torque, in Nm

M_{Tmin} = minimum value of the torque, in Nm

Q_{RFN} = alternating radial force related to the web, in N

R_{H}, R_{G} = fillet radius at junction of web and pin or journal, in mm

S = pin overlap, in mm $S = \frac{D+D_{\mathrm{G}}}{2} - E$

T_{H}, T_{G} = recess of pin or journal fillet radius into web measured from web face, in mm

W = axial thickness of web, in mm

W_{eqw} = section modulus related to cross-section of web, in mm^3

W_{e} = section modulus related to cross-section of axially bored crankpin, in mm^3

W_{p} = polar section modulus related to cross-section of axially bored crankpin or bored journal, in mm^3

y = distance between the adjacent generating lines of journal and pin, in mm

Note: $y \geq 0{,}05D_{\mathrm{S}}$, where y is less than $0{,}1D_{\mathrm{S}}$ special consideration is to be given to the effect of the stress due to the shrink fit on the fatigue strength at the crankpin fillet

α_B = bending stress concentration factor for crankpin fillet

α_T = torsional stress concentration factor for crankpin fillet

β_B = bending stress concentration factor for main journal fillet

Note: α_B and β_B are defined as the ratio of the maximum equivalent stress (von Mises) occurring in the fillets under bending load, to the nominal bending stress related to the web cross-section. *See Figure 1.3.7 Stress concentration factors in crankshaft fillets*.

β_Q = compression stress concentration factor for main journal fillet

Note: β_Q is defined as the ratio of the maximum equivalent stress (von Mises) occurring in the fillet due to the radial force, to the nominal compressive stress related to the web cross-section.

β_T = torsional stress concentration factor for main journal fillet

Note: α_T and β_T are defined as the ratio of the maximum equivalent shear stress occurring in the fillets under torsional load, to the nominal torsional stress related to the axially bored crankpin or journal cross-section. *See Figure 1.3.7 Stress concentration factors in crankshaft fillets*.

γ_B = bending stress concentration factor for outlet of crankpin oil bore

γ_T = torsional stress concentration factor for outlet of crankpin oil bore

Note. γ_B and γ_T are defined as the ratio of the maximum principal stress occurring at the outlet of the crankpin oil-hole under bending and torsional loads respectively, to the corresponding nominal stress related to the axially bored crankpin cross section. *See Figure 1.3.8 Stress concentration factors and stress distribution at the edges of oil drillings.*

σ_{add} = additional bending stress due to misalignment and bedplate deformation as well as due to axial and bending vibrations

σ_B = specified minimum UTS of crankshaft material, in N/mm^2

σ_{BFN} = nominal alternating bending stress related to the web, in N/mm^2

σ_{BG} = alternating bending stress in journal fillet, in N/mm^2

σ_{BH} = alternating bending stress in crankpin fillet, in N/mm^2

σ_{BO} = alternating bending stress in the outlet of the oil bore, in N/mm^2

σ_{BON} = nominal alternating bending stress in the outlet of the oil bore related to the crankpin diameter, in N/mm^2

σ_{QFN} = nominal alternating compressive stress due to radial force related to the web, in N/mm^2

σ_{DW} = allowable fatigue strength of crankshaft, in N/mm^2

σ_{SP} = minimum yield strength of material for journal pin, in N/mm^2

σ_{SW} = minimum yield strength of material for crankweb, in N/mm^2

σ_{TO} = alternating torsional stress in the outlet of the crankpin oil bore, in N/mm

σ_y = equivalent alternating stress for crankpin fillet, journal fillet or outlet of crankpin oil bore as applicable, in N/mm^3

τ_H = alternating torsional stress in crankpin fillet, in N/mm^2

τ_G = alternating torsional stress in journal fillet, in N/mm^2

τ_N = calculated nominal alternating torsional stress referred to crankpin or journal (as applicable), in N/mm^2

τ_a = manufacturer stated crankshaft half range torsional stress limit, in N/mm^2

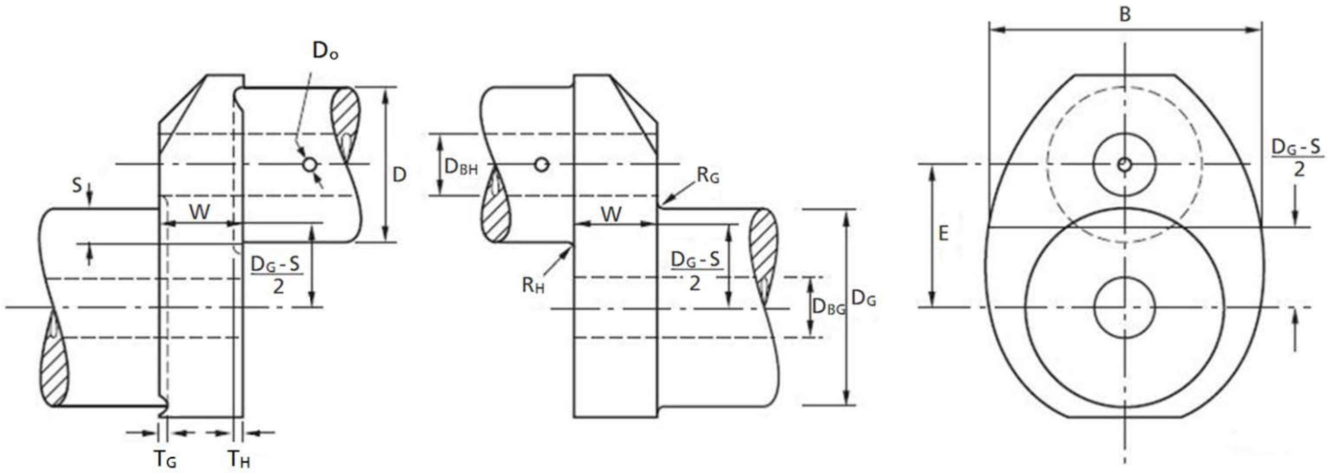


Figure 1.3.3 Crank dimensions for overlapped crankshaft

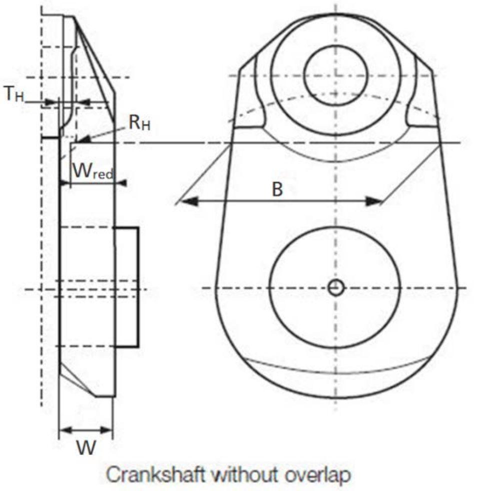


Figure 1.3.4 Crank dimensions for crankshaft without overlap

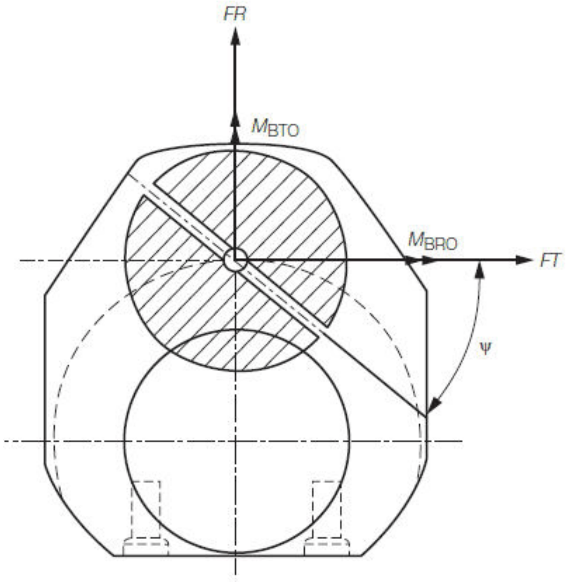


Figure 1.3.5 Crankpin section through the oil bore

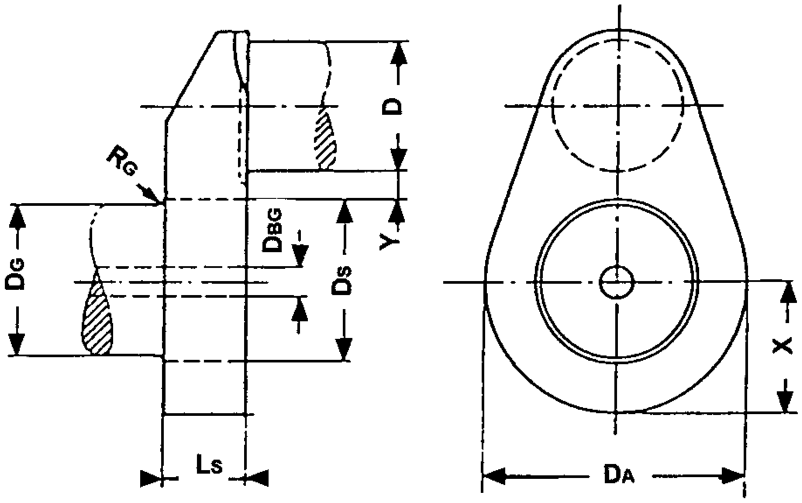
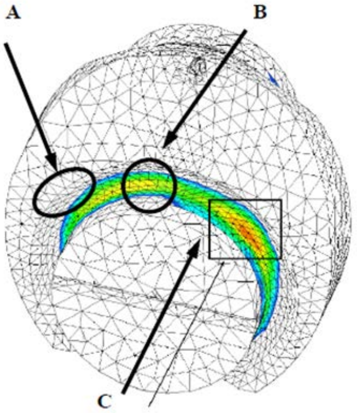


Figure 1.3.6 Crankthrow of semi-built crankshaf



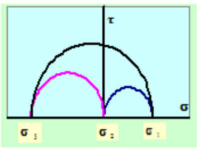
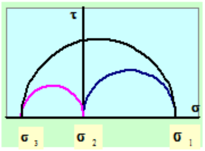
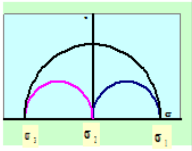
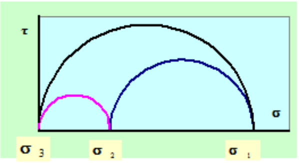
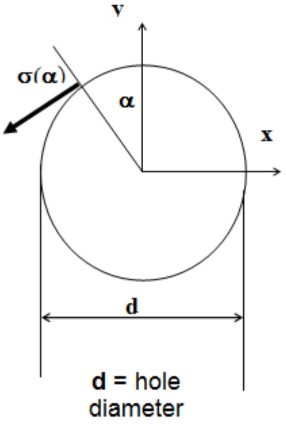
Stress Type	Stress	Max $\|\|\sigma_3\|\|$	Max σ_1	
Torsional loading	Location of maximal stresses	*A*	*C*	*B*
	Typical principal stress system Mohr's circle diagram with $\sigma_2 = 0$	 $\|\|\sigma_3\|\| > \sigma_1$	 $\sigma_1 > \|\|\sigma_3\|\|$	 $\sigma_1 \approx \|\|\sigma_3\|\|$
	Equivalent stress and **S.C.F.**	$\tau_{equiv} = \frac{\sigma_1 - \sigma_3}{2}$ S.C.F. = $\frac{\tau_{equiv}}{\tau_n}$ for α_T, β_T		
Bending loading	Location of maximal stresses	*B*	*B*	*B*
	Typical principal stress system Mohr's circle diagram with $\sigma_3 = 0$			$\sigma_2 \neq 0$
	Equivalent stress and **S.C.F.**	$\sigma equiv = \sqrt{\sigma_1{}^2 + \sigma_2{}^2 - \sigma_1 . \sigma_2}$ S.C.F. $= \frac{\sigma\ equiv}{\sigma_n}$ for $\alpha_B, \beta_B, \beta_Q$		

Figure 1.3.7 Stress concentration factors in crankshaft fillets



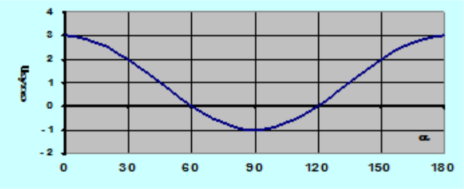
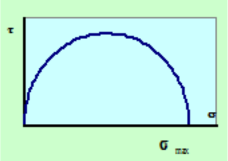
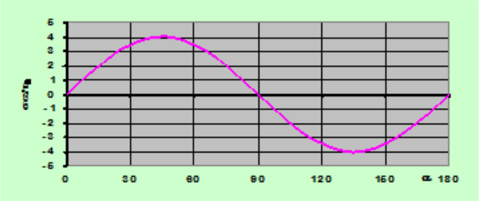
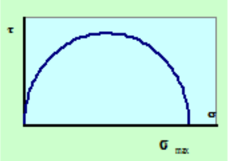
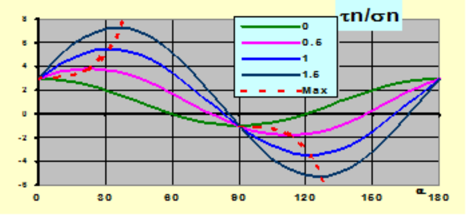
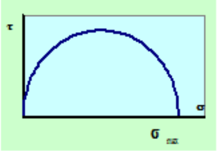
Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma_n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha = \sigma_n \gamma_B / 3\,[1+2\cos(2\alpha)]$ 	 $\gamma_B = \sigma_{max}/\sigma_n$ for $\alpha = k\pi$
Shear	$\begin{bmatrix} 0 & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_T \tau_n \sin(2\alpha)$ 	 $\gamma_T = \sigma_{max}/\tau_n$ for $\alpha = \frac{\pi}{4} + k\frac{\pi}{2}$
Tension + shear	$\begin{bmatrix} \sigma_n & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3}\sigma_n\left\{1+2\left[\cos(2\alpha)+\frac{3}{2}\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\sin(2\alpha)\right]\right\}$ 	 $\sigma_{max} = \frac{\gamma_B}{3}\sigma_n\left[1+2\sqrt{1+\frac{9}{4}\left(\frac{\gamma_T}{\gamma_B}\frac{\tau_n}{\sigma_n}\right)^2}\right]$ for $\alpha = \frac{1}{2}\mathrm{tg}^{-1}\left(\frac{3\gamma_T\tau_n}{2\gamma_B\sigma_n}\right)$

Figure 1.3.8 Stress Concentration Factors and stress distribution at the edges of oil drillings

3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions

3.5.1 The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints (distance L_3, *see Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*).

3.5.2 The bending moments, M_{BR} and M_{BT}, are calculated in the relevant section based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting-rod force, respectively (*see Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows*). For crankthrows with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (*see Figure 1.3.10 Bending moment and shear force for V engine crankthrows*).

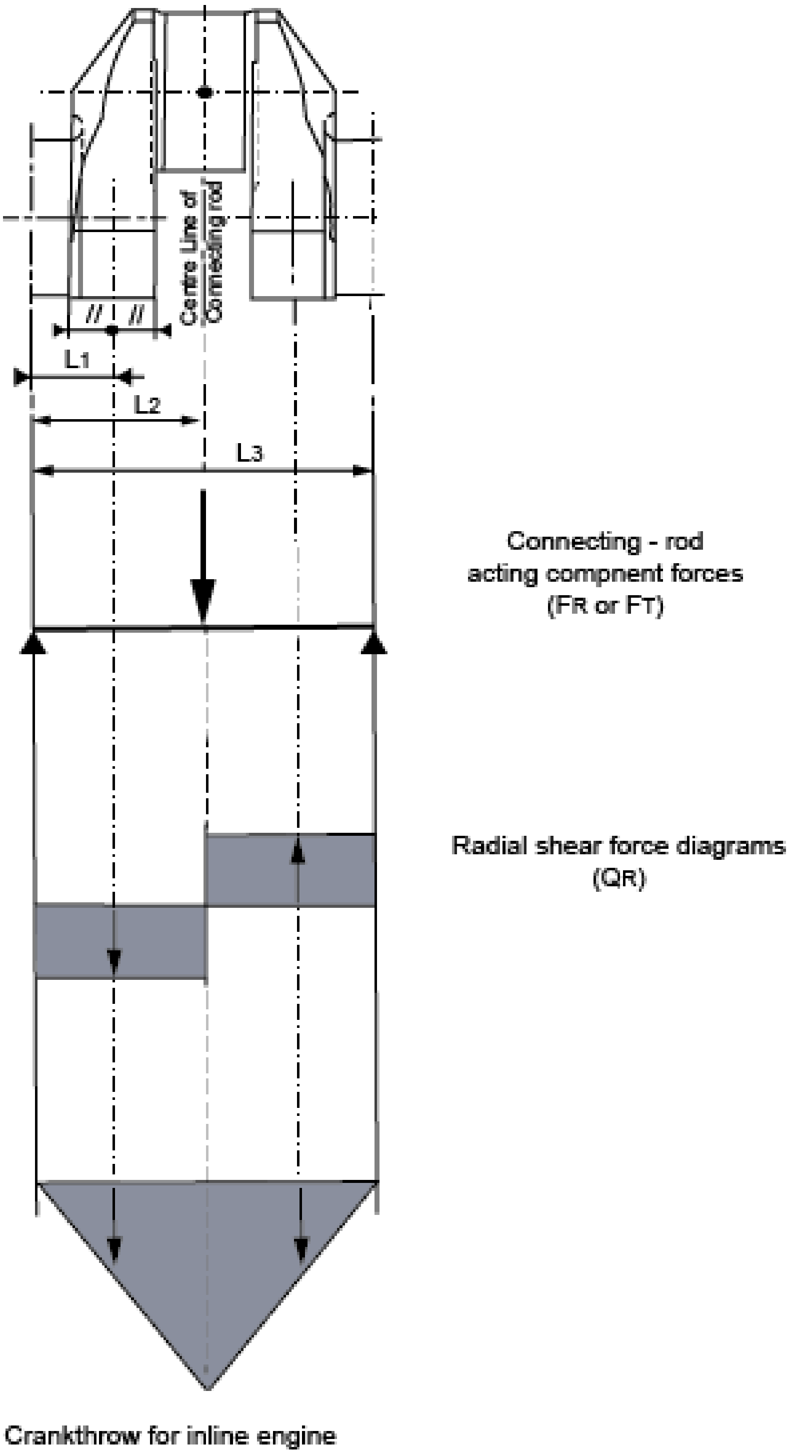

Centre Line of Connecting rod
L1
L2
L3
Connecting - rod acting compnent forces (FR or FT)
Radial shear force diagrams (QR)
Crankthrow for inline engine

Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows

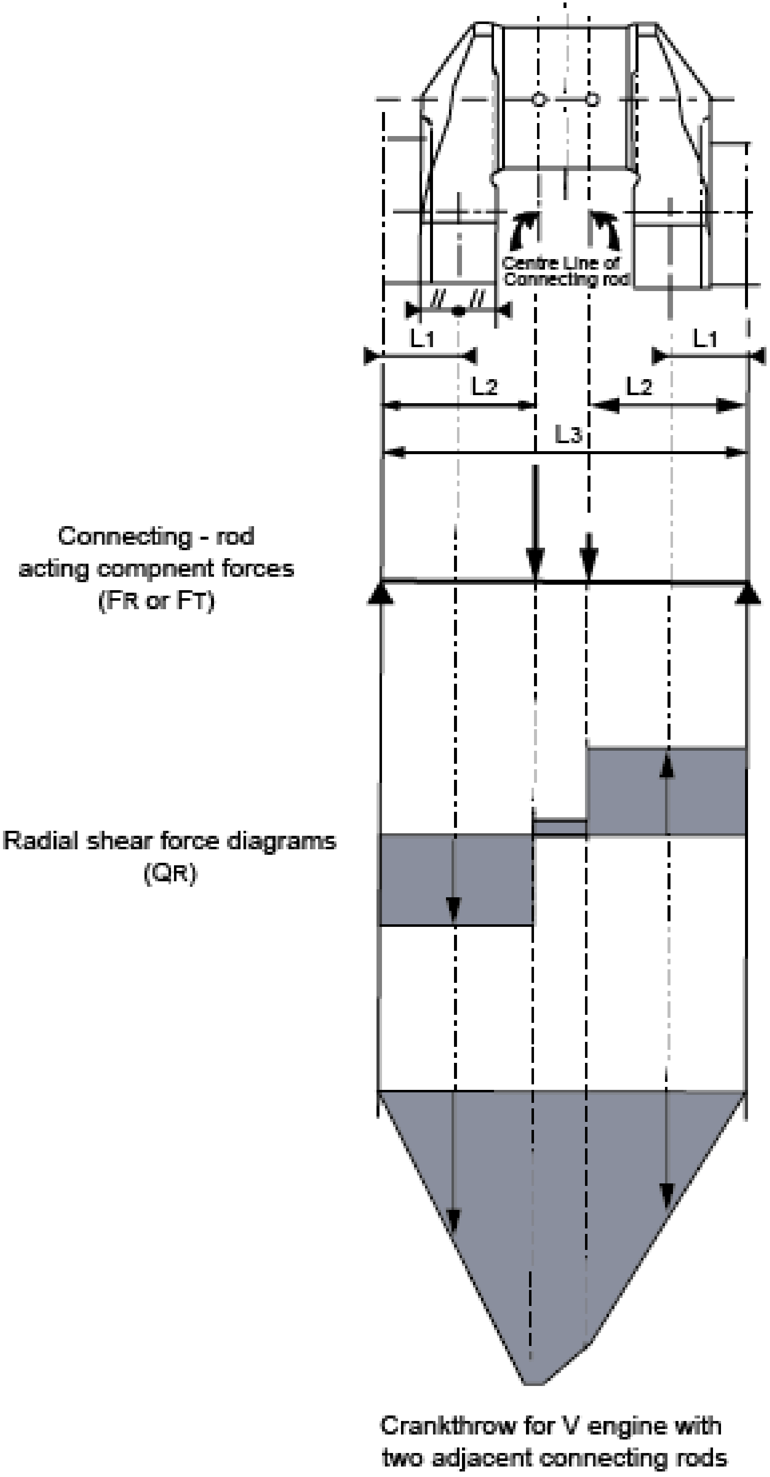

Centre Line of Connecting rod
L1
L1
L2
L2
L3
Connecting - rod acting compnent forces (F_R or F_T)
Radial shear force diagrams (Q_R)
Crankthrow for V engine with two adjacent connecting rods

Figure 1.3.10 Bending moment and shear force for V engine crankthrows

3.5.3 The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of the solid web (distance L_1) and are derived from the radial component of the connecting-rod force. The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crankweb. This reference section results from the web thickness W and the web width B (*see Figure 1.3.3 Crank dimensions for overlapped crankshaft* and *Figure 1.3.4 Crank dimensions for crankshaft without overlap*). Mean stresses are neglected.

3.5.4 The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. *See Figure 1.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 1.3.10 Bending moment and shear force for V engine crankthrows*. M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

3.6 Calculation of bending stresses

3.6.1 The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position are to be calculated over one working cycle. Using the forces calculated over one working cycle and taking into account the distance from the main bearing midpoint, the time curve of the bending moments, M_{BRF}, M_{BRO} and M_{BTO}, and radial forces, Q_{RF},as defined in *Pt 10, Ch 1, 3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions 3.5.2* and *Pt 10, Ch 1, 3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions 3.5.3* are then calculated.

3.6.2 Nominal bending stresses are referred to the web bending modulus.

3.6.3 In case of V-type engines, the bending moments – progressively calculated from the gas and inertia forces – of the two cylinders acting on one crankthrow are superposed according to phase. Different designs (forked connecting-rod, articulated-type connecting-rod or adjacent connecting-rods) shall be taken into account.

3.6.4 Where there are cranks of different geometrical configurations in one crankshaft, the calculation is to cover all crank variants.

3.6.5 The decisive alternating values will then be calculated according to:

$$X_{\mathrm{N}} = \pm \frac{1}{2}(X_{\max} - X_{\min})$$

where

X_N is considered as alternating force, moment or stress

X_{max} is maximum value within one working cycle

X_{min} is minimum value within one working cycle

3.6.6 Nominal alternating bending and compressive stresses in web cross-section are calculated as follows:

$$\sigma_{\mathrm{BFN}} = \pm \frac{M_{\mathrm{BRFN}}}{W_{\mathrm{eqw}}} \cdot 10^3 \cdot K_{\mathrm{e}} \text{ N/mm}^2$$

$$\sigma_{\mathrm{QFN}} = \pm \frac{Q_{\mathrm{RFN}}}{F} \cdot K_{\mathrm{e}} \text{ N/mm}^2$$

where

$M_{\mathrm{BRFN}} = \pm \frac{1}{2}(M_{\mathrm{BRFmax}} - M_{\mathrm{BRFmin}})$ Nm

$W_{\mathrm{eqw}} = \frac{BW^2}{6}$ mm^3

K_e = 0,8 for crosshead engines
= 1,0 for trunk piston engines

$Q_{\mathrm{RFN}} = \pm \frac{1}{2}(Q_{\mathrm{RFmax}} - Q_{\mathrm{RFmin}})$ N

$F = BW$ mm^2

3.6.7 Nominal alternating bending stress in the outlet of the crankpin oil bore is calculated as follows:

$$\sigma_{\mathrm{BON}} = \pm \frac{M_{\mathrm{BON}}}{W_{\mathrm{e}}} \cdot 10^3 \text{ N/mm}^2$$

where

M_{BON} is taken as the half range value $M_{BON} = \pm 0{,}5(M_{BOmax} - M_{BOmin})$

and

$M_{BO} = (M_{BTO} \cos\psi + M_{BRO} \sin\psi)$, ψ = angular position in degrees, *see Figure 1.3.5 Crankpin section through the oil bore*

M_{BRO} = bending moment of the radial component of the connecting-rod force

M_{BTO} = bending moment of the tangential component of the connecting-rod force

$$W_{\mathrm{e}} = \frac{\pi}{32}\left(\frac{D^4 - {D_{\mathrm{BH}}}^4}{D}\right) \text{ mm}^3$$

3.6.8 Alternating bending stresses for the crankpin fillet and journal fillet are calculated as follows:

(a) For the crankpin fillet:

$$\sigma_{\mathrm{BH}} = \pm(\alpha_{\mathrm{b}} \sigma_{\mathrm{BFN}}) \text{ N/mm}^2$$

where

α_B is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.5 (a)*

(b) For the journal fillet:

$$\sigma_{\mathrm{BG}} = \pm(\beta_{\mathrm{B}} \sigma_{\mathrm{BFN}} + \beta_{\mathrm{Q}} \sigma_{\mathrm{QFN}}) \text{ N/mm}^2$$

where

β_B is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.6 (a)*

β_Q is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.6 (b)*

3.6.9 Alternating bending stresses for the outlet of crankpin oil bore are calculated as follows:

$\sigma_{\mathrm{BO}} = \pm(\gamma_{\mathrm{B}}\sigma_{\mathrm{BON}})$ N/mm
where
γ_{B} is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.7 (a).*

3.7 Calculation of torsional stresses

3.7.1 The nominal alternating torsional stress, τ_{N}, is to be taken into consideration. The value is to be derived from forced–damped vibration calculations of the complete dynamic system. Alternative methods will be given consideration. The engine designer is to advise the maximum level of alternating vibratory stress that is permitted.

3.7.2 τ_{a} or τ_{N} (as applicable) is to be applied as a limiting value for the torsional vibration assessment required by Pt 13, Ch 1, Torsional Vibration.

3.7.3 Nominal alternating torsional stress is calculated as follows:

$$\tau_{\mathrm{N}} = \pm\frac{M_{\mathrm{TN}}}{W_{\mathrm{P}}}\cdot 10^3 \text{ N/mm}^2$$

where

$$M_{\mathrm{TN}} = \pm\frac{1}{2}(M_{\mathrm{Tmax}} - M_{\mathrm{Tmin}}) \text{ Nm}$$

$W_{\mathrm{p}} = \frac{\pi}{16}\left(\frac{D^4 - D_{\mathrm{BH}}{}^4}{D}\right)$ mm^3 for the crankpin, or $W_{\mathrm{p}} = \frac{\pi}{16}\left(\frac{D_{\mathrm{G}}{}^4 - D_{\mathrm{BG}}{}^4}{D_{\mathrm{G}}}\right)$ mm^3 for the journal

τ_N is to be ascertained from assessment of the torsional vibration calculations where the maximum and minimum torques are determined for every mass point of the complete dynamic system and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the first order up to and including the 15th order for 2-stroke cycle engines and from the 0,5th order up to and including the 12th order for 4-stroke cycle engines. While doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring in one of the cylinders when no combustion occurs but only compression cycle). The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected.

3.7.4 For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in calculations is to be the highest calculated value, according to the method described in *Pt 10, Ch 1, 3.7 Calculation of torsional stress 3.7.3*, occurring at the most torsionally loaded mass point of the crankshaft system.

3.7.5 The approval of the crankshaft will be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by the engine manufacturer). For each installation it is to be ensured by calculation that the maximum approved nominal alternating torsional stress is not exceeded. *See Pt 13, Ch 1 Torsional vibration*.

3.7.6 Alternating torsional stresses for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore are calculated as follows.

(a) Maximum alternating torsional stress in crankpin fillet:

$\tau_{\mathrm{H}} = \pm(\alpha_{\mathrm{T}}\tau_{\mathrm{N}})$ N/mm^2
where
α_{T} is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.5 (b)*

(b) Maximum alternating torsional stress in the journal fillet (not applicable to semi-built crankshafts):
$\tau_{\mathrm{G}} = \pm(\beta_{\mathrm{T}}\tau_{\mathrm{N}})$ N/mm^2
where
β_{T} is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.6 (c)*

(c) Maximum alternating torsional stress in the outlet of the crankpin oil bore:
$\sigma_{\mathrm{TO}} = \pm(\gamma_{\mathrm{T}}\tau_{\mathrm{N}})$ N/mm^2
where
γ_{T} is calculated according to *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.7 (b).*

3.8 Stress concentration factors

3.8.1 Stress concentration factors (SCF) are to be calculated using the analytical formulae outlined in this Section.

3.8.2 Crankshaft variables to be used in calculating the geometric SCF are shown in *Table 1.3.1 Crankshaft variables for SCF calculation*, their limits of applicability are shown in *Table 1.3.2 Crankshaft variable boundaries for analytical SCF calculation*.

3.8.3 Where the geometry of the crankshaft is outside the boundaries (*see Table 1.3.2 Crankshaft variable boundaries for analytical SCF calculation*) of the analytical SCF the calculation method detailed in Chapter 1 of LR *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* may be undertaken.

3.8.4 Where reliable experimental measurements and/or calculations are available, which can allow direct assessment of SCF, these can be used. The relevant documents and their analysis are to be submitted for consideration in order to demonstrate their equivalence. This is always to be performed when dimensions are outside the boundaries shown in *Table 1.3.2 Crankshaft variable boundaries for analytical SCF calculation*.

3.8.5 Chapters 1 and 3 of LR *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* describe how finite element (FE) analyses can be used for the calculation of the SCF. Care needs to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

Table 1.3.1 Crankshaft variables for SCF calculation

Variable	Function
r	$= R_H/D$ for crankpin fillet $= R_G/D$ for journal fillet
s	$= S/D$
w	$= W/D$ crankshafts with overlap $= W_{red}/D$ crankshafts without overlap
b	$= B/D$
d_o	$= D_O/D$
d_G	$= D_{BG}/D$
d_H	$= D_{BH}/D$
t_H	$= T_H/D$
t_G	$= T_G/D$

Table 1.3.2 Crankshaft variable boundaries for analytical SCF calculation

Lower bound	Variable	Upper bound
	s	$\leq 0{,}5$
$0{,}2 \leq$	w	$\leq 0{,}8$
$1{,}1 \leq$	b	$\leq 2{,}2$
$0{,}03 \leq$	r	$\leq 0{,}13$
$0 \leq$	d_G	$\leq 0{,}8$
$0 \leq$	d_H	$\leq 0{,}8$
$0 \leq$	d_o	$\leq 0{,}2$

Notes

Low range of *s* can be extended down to large negative values provided that:

- If calculated *f*(recess) < 1 then the factor *f*(recess) is not to be considered (*f*(recess) = 1)
- If $s < -0{,}5$ then $f(s,w)$ and $f(r,s)$ are to be evaluated replacing actual value of *s* by –0,5.

3.8.6 Crankpin SCF are calculated as follows:

(a) Bending

$$\alpha_B = 2{,}6914 f(s,w).f(w).f(b).f(r).f(d_G).f(d_H).f(\text{recess})$$

where

$$f(s,w) = -4{,}1883 + 29{,}2004w - 77{,}5925w^2 + 91{,}9454w^3 - 40{,}0416w^4 + (1 - s)(9{,}5440 - 58{,}3480w + 159{,}3415w^2 - 192{,}5846w^3 + 85{,}2916w^4) + (1 - s)^2(-3{,}8399 + 25{,}0444w - 70{,}5571w^2 + 87{,}0328w^3 - 39{,}1832w^4)$$

$$f(w) = 2{,}1790w^{0{,}7171}$$

$$f(b) = 0{,}684 - 0{,}0077b + 0{,}1473b^2$$

$$f(r) = 0{,}2081r^{(-0{,}5231)}$$

$$f(d_G) = 0{,}9993 + 0{,}27d_G - 1{,}0211d_G^2 + 0{,}5306d_G^3$$

$$f(d_H) = 0{,}9978 + 0{,}3145d_H - 1{,}5241d_H^2 + 2{,}4147d_H^3$$

$$f(\text{recess}) = 1 + (t_H + t_G)(1{,}8 + 3{,}2s)$$

(b) Torsion

$$\alpha_T = 0{,}8f(r,s).f(b).f(w)$$

where

$$f(r,s) = r^{(-0{,}322 + 0{,}1015(1-s))}$$

$$f(b) = 7{,}8955 - 10{,}654b + 5{,}3482b^2 - 0{,}857b^3$$

$$f(w) = w^{(-0{,}145)}$$

3.8.7 Journal fillet SCF are calculated as follows (not applicable to semi-built crankshafts):

(a) Bending

$\beta_B = 2{,}7146 f_B(s,w).f_B(w).f_B(b).f_B(r).f_B(d_G).f_B(d_H).f(\text{recess})$

where

$f_B(s,w) = -1{,}7625 + 2{,}9821w - 1{,}5276w^2 + (1 - s)(5{,}1169 - 5{,}8089w + 3{,}1391w^2) + (1 - s)^2(-2{,}1567 + 2{,}3297w - 1{,}2952w^2)$

$f_B(w) = 2{,}2422w^{0{,}7548}$

$f_B(b) = 0{,}5616 + 0{,}1197b + 0{,}1176b^2$

$f_B(r) = 0{,}1908r^{(-0{,}5568)}$

$f_B(d_G) = 1{,}0012 - 0{,}6441d_G + 1{,}2265d_G^2$

$f_B(d_H) = 1{,}0022 - 0{,}1903d_H + 0{,}0073d_H^2$

$f(\text{recess}) = 1 + (t_H + t_G)(1{,}8 + 3{,}2s)$

(b) Compression due to the radial force:

$\beta_Q = 3{,}0128 f_Q(s).f_Q(w).f_Q(b).f_Q(r).f_Q(d_H).f(\text{recess})$

where

$f_Q(s) = 0{,}4368 + 2{,}1630(1 - s) - 1{,}5212(1 - s)^2$

$f_Q(w) = \frac{w}{0{,}0637+0{,}9369w}$

$f_Q(b) = b - 0{,}5$

$f_Q(r) = 0{,}5331r^{(-0{,}2038)}$

$f_Q(d_H) = 0{,}9937 - 1{,}1949d_H + 1{,}7373d_H^2$

$f(\text{recess}) = 1 + (t_H + t_G)(1{,}8 + 3{,}2s)$

(c) Torsion:

$\beta_T = \alpha_T$ if the diameters and fillet radii of crankpin and journal are the same, or

$\beta_T = 0{,}8f(r,s).f(b).f(w)$ if crankpin and journal diameters and/or radii are of different sizes

where

$f(r,s)$, $f(b)$ and $f(w)$ are to be determined in accordance with *Pt 10, Ch 1, 3.8 Stress concentration factors 3.8.6 (b)*, however, the radius of the journal fillet is to be related to the journal diameter: $r = \frac{R_G}{D_G}$

3.8.8 Crankpin oil bore SCF for radially drilled oil holes are calculated as follows:

(a) Bending

$\gamma_B = 3 - 5{,}88d_o + 34{,}6{d_o}^2$

(b) Torsion

$\gamma_T = 4 - 6d_o + 30{d_o}^2$

3.9 Additional bending stress

3.9.1 In addition to the alternating bending stresses in fillets (*see Pt 10, Ch 1, 3.6 Calculation of bending stresses 3.6.8*) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying α_{add} as given by *Table 1.3.3 Additional bending stresses*.

Table 1.3.3 Additional bending stresses

<table>
<tr><th>Type of engine</th><th>σ_{add}</th></tr>
<tr><td>Crosshead engines</td><td>±30 N/mm² (*see* Note 1)</td></tr>
<tr><td>Trunk piston engines</td><td>±10 N/mm²</td></tr>
<tr><td colspan="2">Note 1. The additional stress of ±30 N/mm² is composed of two components:
(a) an additional stress of ±20 N/mm² resulting from axial vibration
(b) an additional stress of ±10 N/mm² resulting from misalignment/bedplate deformation.</td></tr>
</table>

3.9.2 It is recommended that a value of ±20 N/mm^2 be used for the axial vibration component for assessment purposes where axial vibration calculation results of the complete dynamic system (engine/shafting/gearing/propeller) are not available. Where axial vibration calculation results of the complete dynamic system are available, the calculated figures can be used instead.

3.10 Equivalent alternating stress

3.10.1 In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (*see Figure 1.3.7 Stress concentration factors in crankshaft fillets*). As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the von Mises criterion.

3.10.2 At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (*see Figure 1.3.8 Stress concentration factors and stress distribution at the edges of oil drillings*).

3.10.3 The above two different ways of equivalent stress evaluation both lead to stresses which can be compared to the same fatigue strength value of crankshaft assessed according to the von Mises criterion.

3.10.4 Equivalent alternating stress, σ_v, is defined as:

(a) For the crankpin fillet:

$$\sigma_{\mathrm{v}} = \pm\sqrt{(\sigma_{\mathrm{BG}} + \sigma_{\mathrm{add}})^2 + 3\tau_{\mathrm{G}}^2} \text{ N/mm}^2$$

(b) For the journal fillet:

$$\sigma_{\mathrm{v}} = \pm\sqrt{(\sigma_{\mathrm{BH}} + \sigma_{\mathrm{add}})^2 + 3\tau_{\mathrm{H}}^2} \text{ N/mm}^2$$

(c) For the outlet of crankpin oil bore:

$$\sigma_{\mathrm{v}} = \pm\tfrac{1}{3}\sigma_{\mathrm{BO}}\left[1 + 2\sqrt{1 + \tfrac{9}{4}\left(\frac{\sigma_{\mathrm{TO}}}{\sigma_{\mathrm{BO}}}\right)^2}\right] \text{ N/mm}^2$$

3.11 Fatigue strength

3.11.1 The fatigue strength is to be understood as that value of equivalent alternating stress (von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength can be evaluated by means of the following formulae.

(a) Related to the crankpin diameter:

$$\sigma_{\mathrm{DW}} = \pm K(0{,}42\sigma_{\mathrm{B}} + 39{,}3)\left[0{,}264 + 1{,}073D^{-0.2} + \frac{785-\sigma_{\mathrm{B}}}{4900} + \frac{196}{\sigma_{\mathrm{B}}}\sqrt{\frac{1}{R_{\mathrm{X}}}}\right] \text{ N/mm}^2$$

with

$R_X = R_H$ in the fillet area

$R_X = D_o/2$ in the oil bore area

(b) Related to the journal diameter:

$$\sigma_{\mathrm{DW}} = \pm K(0{,}42\sigma_{\mathrm{B}} + 39{,}3)\left[0{,}264 + 1{,}073{D_{\mathrm{G}}}^{-0.2} + \frac{785-\sigma_{\mathrm{B}}}{4900} + \frac{196}{\sigma_{\mathrm{B}}}\sqrt{\frac{1}{R_{\mathrm{G}}}}\right] \text{ N/mm}^2$$

where

$K = K_1 K_2$

K_1 = fatigue endurance factor appropriate to the manufacturing process
= 1,05 for continuous grain flow forged or drop-forged crankshafts
= 1,0 for free form forged crankshafts (without continuous grain flow)
= 0,93 for cast steel crankshafts with cold rolling treatment in fillet area manufactured by companies using a LR approved cold rolling process

K_2 = fatigue enhancement factor for surface treatment. These treatments are to be applied to the fillet radii

A value for K_2 will be assigned upon application by the engine designers. Full details of the process, together with the results of full scale fatigue tests will be required to be submitted for consideration *See LR Guidance note - Guidance for the evaluation of Crankshaft Fatigue Tests*. Alternatively, the following values may be taken (surface hardened zone to include fillet radii):

K_2 = 1,15 for induction hardened
= 1,25 for nitrided

Where a value of K_1 or K_2 greater than unity is to be applied then details of the manufacturing process are to be submitted. An enhanced K_1 factor will be considered, subject to special approval of the manufacturing specification. *See Materials and Qualification Procedures for Ships, Book E, Procedure MQPS 5-2*.

3.11.2 The formulae in *Pt 10, Ch 1, 3.11 Fatigue strength 3.11.1* are subject to geometry limits. The junction of the oil hole with the crankpin or main journal surface is to be formed with an adequate radius and smooth surface finish down to a minimum depth equal to 1,5 times the oil bore diameter and for calculation purposes R_H, R_G or R_X are to be taken as not less than 2 mm.

3.11.3 Fatigue strength calculations or alternatively fatigue test results determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow may be required to demonstrate acceptability. The experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment are to be submitted for approval by LR. The procedure is to include as a minimum: method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, and confidence number. *See also* LR *Guidance for the Evaluation of Crankshaft Fatigue Tests.*

3.11.4 When journal diameter is equal or larger than the crankpin diameter, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate fatigue strength calculations or, alternatively, fatigue test results may be required.

3.11.5 Only surface treatment processes approved by LR are permitted. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Chapter 2 of the LR *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method*.

3.12 Acceptability criteria

3.12.1 The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. The acceptability factor, *Q*, is to be greater than or equal to 1,15 for the crankpin fillet, the journal fillet, the outlet of crankpin oil bore:

$$Q = \frac{\sigma_{\mathrm{DW}}}{\sigma_{\mathrm{v}}}$$

3.13 Shrink fit of semi-built crankshafts

3.13.1 The following formulae are applicable to crankshafts assembled by shrinking main journals into the crankwebs, *see also Figure 1.3.6 Crankthrow of semi-built crankshaft*.

3.13.2 In general, the radius of transition, R_G, between the main journal diameter, D_G, and the shrink diameter, D_S, is to be not less than $0{,}015D_G$ or $0{,}5(D_S – D_G)$ where the greater value is to be considered.

3.13.3 Deviations from these parameters will be specially considered.

3.13.4 The maximum permissible internal diameter in the journal pin is to be calculated in accordance with the following formula, this condition serves to avoid plasticity in the hole of the journal pin:

$$D_{\mathrm{BG}} = D_{\mathrm{S}} \sqrt{1 - \frac{4000 S_{\mathrm{R}} M_{\max}}{\mu \pi D_{\mathrm{S}}^2 L_{\mathrm{S}} \sigma_{\mathrm{SP}}}} \text{ mm}$$

where

S_R = safety factor against slipping; however, a value not less than 2 is to be taken unless documented by experiments.

M_{max} = absolute maximum value of the torque M_{Tmax} in accordance with *Pt 10, Ch 1, 3.6 Calculation of torsional stresses 3.6.3*, in Nm

μ = coefficient for static friction; however, a value not greater than 0,2 is to be taken unless documented by experiments.

3.13.5 The actual oversize *Z* of the shrink fit must be within the limits Z_{min} and Z_{max} calculated in accordance *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts 3.13.7* and *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts 3.13.8.* When *Pt 10, Ch 1, 3.13 Shrink fit of semi-built crankshafts 3.13.5* cannot be complied with, then the calculated values of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such cases Z_{min} and Z_{max} are to be established from FEM calculations.

3.13.6 The minimum required diametral Interference is to be taken as the greater of:

$$Z_{\min} \geq \frac{\sigma_{\mathrm{SW}} D_{\mathrm{S}}}{E_{\mathrm{m}}} \text{ mm}$$

and

$$Z_{\min} \geq \frac{4000}{\mu\pi} \; \frac{S_{\mathrm{R}} M_{\max}}{E_{\mathrm{m}} D_{\mathrm{S}} L_{\mathrm{S}}} \; \frac{1 - {Q_{\mathrm{A}}}^2 {Q_{\mathrm{S}}}^2}{(1 - {Q_{\mathrm{A}}}^2)(1 - {Q_{\mathrm{S}}}^2)} \text{ mm}$$

where

Q_A = web ratio, $Q_{\mathrm{A}} = \dfrac{D_{\mathrm{S}}}{D_{\mathrm{A}}}$

Q_S = shaft ratio, $Q_{\mathrm{S}} = \dfrac{D_{\mathrm{BG}}}{D_{\mathrm{S}}}$

3.13.7 The maximum diametral interference is not to be greater than:

$$Z_{\max} \leq D_{\mathrm{S}} \left(\frac{\sigma_{\mathrm{SW}}}{E_{\mathrm{m}}} + \frac{0{,}8}{1000} \right) \text{ mm}$$

This condition serves to restrict the shrinkage induced mean stress in the fillet.

3.13.8 Reference marks are to be provided on the outer junction of the crankwebs with the journals.

■ *Section 7*
Control and monitoring of main, auxiliary and emergency engines

7.3 Auxiliary engine governors

7.3.2 If an engine cannot achieve the requirements of *Pt 10, Ch 1, 7.3 Auxiliary engine governors 7.3.1* then the actual load step is to be declared and verified through testing to ensure the requirements specified in *Pt 16, Ch 2, ~~1.7 Quality of power supplies~~ 1.8 Quality of power supplies* are satisfied. In cases where a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10 per cent of the rated speed is acceptable, provided this does not cause the intervention of the overspeed device as required by *Pt 10, Ch 1, 7.4 Overspeed protective devices 7.4.1*.

7.3.3 Emergency engines are to comply with *Pt 10, Ch 1, 7.3 Auxiliary engine governors 7.3.1* except that the initial load required by *Pt 10, Ch 1, 7.3 Auxiliary engine governors 7.3.1.(b)* is to be not less than the total connected emergency statutory load, or if their total consumer load is applied in steps, the following requirements are to be met:
(a) the total load is supplied within 45 seconds from power failure on the main switchboard;
(b) the maximum step load is declared and demonstrated; and
(c) the power distribution system is designed such that the declared maximum step loading is not exceeded.

7.3.5 For alternating current installations, the permanent speed variation of the machines intended for parallel operation are to be equal within a tolerance of ± 0,5 per cent. Momentary speed variations with load changes in accordance with *Pt 10, Ch 1, 7.3 Auxiliary engine governors 7.3.1* are to return to and remain within one per cent of the final steady state speed. This should normally be accomplished within five but in no case more than eight seconds. For quality of power supplies, *see Pt 16, Ch 2, ~~1.7 Design and construction~~ 1.8 Quality of power supplies*.

7.8 Emergency engines

(Part only shown)
Table 1.7.4 Emergency engines: Alarms and safeguards

Item	Alarm for engine power <220kW	Alarm for engine power ≥ 220 kW	Note

■ *Section 8*
Piping

8.1 Fuel oil, hydraulic and high pressure oil systems

8.1.1 Fuel oil and hydraulic oil piping system arrangements are to comply with ~~*Pt 15 Piping Systems and Pressure Plant*~~ *Pt 15, Ch 1 Piping Design Requirements* and *Pt 15, Ch 3 Machinery Piping Systems* as applicable.

8.1.3 Where fuel oil and hydraulic oil pressure pumps are fitted, and these are essential for engine operation, not less than two fuel oil and two hydraulic oil pressure pumps are to be provided and arranged such that failure of one pump does not render the other pump(s) inoperative. Each fuel oil pump and hydraulic oil pump is to be capable of supplying the quantity of oil for engine operation at its maximum continuous rating and arranged ready for immediate use. ~~Small craft not required to comply with the *HSC Code - International Code of Safety for High Speed Craft, 1994 – Resolution MSC.36(63)* are to be capable of maintaining adequate manoeuvring capability.~~

Paragraphs 8.1.10 and 8.1.11 have been moved to new sub-Section 8.2. Existing paragraph 8.1.12 has been renumbered 8.1.10.

8.2 Additional requirements for fuel oil, hydraulic and high pressure oil systems for special service craft

8.2.1 Small craft not required to comply with the *HSC Code - International Code of Safety for High Speed Craft, 1994 – Resolution MSC.36(63)* are to be capable of maintaining adequate manoeuvring capability.

~~8.1.10~~ 8.2.2 Where multi-engine installations on board special service craft required to comply with the *SOLAS - International Convention for the Safety of Life at Sea* and yachts that are of 500 gt or more are supplied from the same fuel source, means of isolating the fuel supply and spill piping to individual engines is to be provided. These means of isolation are not to affect the operation of the other engines and are to be operable from a position not rendered inaccessible by a fire on any of the engines.

~~8.1.11~~ 8.2.3 Where fuel treatments, additives or emulsified fuel are used as a means of abating exhaust emissions, engines and fuel systems are to be compatible with such additives, treatments and emulsified fuel.

Existing sub-Sections 8.2 to 8.4 have been renumbered 8.3 to 8.5.

Note that this excludes existing paragraphs 8.2.10 to 8.2.12 which have been renumbered 8.6.3 to 8.6.5 as shown below.

~~8.5~~ 8.6 Emissions abatement systems

Existing paragraphs 8.5.1 and 8.5.2 have been renumbered 8.6.1 and 8.6.2.

~~8.2.10~~ 8.6.3 Where emissions abatement equipment is to be installed as part of the exhaust system, the resulting exhaust gas system back pressure is to remain within the allowable limits stated by engine manufacturer, and a means of measuring differential pressure across the emissions abatement equipment is to be provided.

~~8.2.11~~ 8.6.4 Where emissions abatement equipment is installed as part of the exhaust system, the exhaust system is to be capable of safely transmitting the minimum and maximum exhaust gas flows also with the emissions abatement equipment out of operation.

~~8.2.12~~ 8.6.5 Where chemicals or substances are injected into the exhaust gas flow before engine turbo-charger(s) or emissions abatement equipment, this is not to present a risk of damage, chemical attack or performance degradation to the turbo-charger(s) or emissions abatement equipment.

■ *Section 9*
Starting arrangements

9.4 Additional requirements for electric starting for non-SOLAS cargo vessels

~~9.3.6~~ 9.4.1 For cargo ships of less than 500 gross tons which are not required to comply with the *International Convention for the Safety of Life at Sea, 1974*, as amended (SOLAS 74), the emergency source of electrical power may be used as one of the sources of energy required by *Pt 5, Ch 2, 9.3 Electric starting 9.3.1* or *Pt 5, Ch 2, 9.3 Electric starting 9.3.2* for electric starting. Where the emergency source of electrical power is an accumulator battery and it is to be used for electric starting, it is to have the additional capacity required to ensure emergency supplies are not compromised and is to be adequately protected and suitably located for use in an emergency.

Existing sub-Sections 9.4 and 9.5 have been renumbered 9.5 and 9.6.

■ *Section 11*
Factory Acceptance Test and Shipboard Trials of Engines

11.3 Works trials (factory acceptance test)

(Part only shown)

Table 1.11.1 Scope of works trials for engines

Main engines driving propellers and waterjets		
Testing of governor and independent overspeed protective device	—	*See Pt 10, Ch 1, ~~7.2 Main engine governers~~ 7 Control and monitoring of main, auxiliary and emergency engines*
Shutdown device	—	*See Pt 10, Ch 1, ~~7.4 Overspeed protective devices~~ 7 Control and monitoring of main, auxiliary and emergency engines*
Engines driving mechanical auxiliaries		
Trial condition	Duration	Note

11.4 Shipboard trials

(Part only shown)

Table 1.11.2 Scope of ~~on board~~ shipboard trials for ~~diesel~~ engines

Main engines driving fixed-pitch propeller or waterjet (*see* Note 1)		
Trial condition	Duration	Note
Starting and reversing manoeuvres	–	*See Pt 5, Ch 2, 9 Starting arrangements* and *Pt 5, Ch 2, 13 Air compressors* (*see* Note 5)
Note 1. For main propulsion engines driving reversing gears, the tests for main engines driving fixed-pitch propellers apply as appropriate. **Note** 5. Starting manoeuvres are to be carried out in order to verify the capacity of the starting media. The ability of reversible engines to be operated in the reverse direction is to be demonstrated. *See Pt 9, Ch 1, 4.7 Astern power*.		

Part 16, Chapter 2
Electrical Engineering

■ *Section 1*
General requirements

1.4 Surveys

1.4.5 Alternative approach for product assurance;

(a) LR will be prepared to give consideration to the adoption of an approach for product assurance, utilising regular and systematic audits of an organisation's arrangements for assuring product quality, as an alternative to the direct survey of individual items.

(b) Alternative approaches for product assurance are to be approved by LR. In order to obtain approval, the requirements of *Pt 5, Ch 1, 6 Quality Assurance Scheme for Machinery* or *Ch 1, 2.4 Materials Quality Scheme* of the *Rules for Manufacture, Testing and Certification of Materials, July 2017, incorporating Notice No. 1 & 2* are to be complied with. Proposals for equivalent approaches are to be submitted for consideration.

■ *Section 9*
Rotating machines

9.1 General requirements

9.1.4 All machines of 100 kW and over, intended for essential services, are to be surveyed by the Surveyor during manufacture and test, *see also Pt 16, Ch 2, 1.4 Surveys 1.4.5*.

9.1.11 For ~~the HV~~ high voltage rotating machines used for essential services are to comply with the relevant requirements of *Pt 6, Ch 2, 1 General requirements* and *Pt6, Ch2, 9 Rotating machines* of the *Rules and Regulations for the Classification of Ships, July 2017, incorporating Notice No. 1, 2, 3 & 4*.

■ *Section 11*
Electric cables, optical fibre cables and busbar trunking systems (busways)

11.1 General

11.1.4 Electric cables for electric propulsion systems are to be Type Approved in accordance with *LR's Type Approval System Test Specification Number 3* or, alternatively, surveyed by the Surveyors during manufacture and testing to assess compliance with the applicable International or National Standards and application of an acceptable quality management system, *see also Pt 16, Ch 2, 1.4 Surveys 1.4.5*.

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